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IMPROVED DESIGN OF AIR FLOW FOR A TWO STROKE INTERNAL COMBUSTION ENGINE WITHOUT SCAVENGING PROBLEMS TO PROMOTE CLEANER COMBUSTION

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ABSTRACT

In the present work, an attempt is made to reduce scavenging problems by developing a new model of two-stroke spark-ignition engine. This model allows the flow of fresh air through intake valves positioned at the bottom of the cylinder and exit of burnt gases through exhaust ports situated at the top of the cylinder. The exhaust ports are closed by the piston as it moves towards the bottom of the cylinder following which gasoline is injected minimizing the possibility of mixing of fuel with outgoing exhaust gases. During the expansion, the piston unravels the exhaust ports and the burnt gases escape to the atmosphere due to pressure difference. Due to low density at high temperatures, the exhaust gases naturally rise against gravity, reducing the possibility of mixing with incoming fresh air. Further, the investigation on fuel distribution inside the cylinder showed a better distribution during injection against gravity, which promotes cleaner combustion. The combustion analysis was done using Diesel-RK software and flow analysis was done using ANSYS FLUENT.

KEYWORDS: Two Stroke Engine, SI Engine, Scavenging, Intake Valves and Exhaust Ports & Conversion of 4 Stroke to 2 Stroke

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Original Article

1. INTRODUCTION

The effective air flow system in two stroke engines is governed by the scavenging process, which allows the engine to breathe in fresh air and expelling burnt gases after combustion. The incoming air is used to clean out or scavenge the exhaust gases and then to fill or charge the space with fresh air [1]. In conventional engines, scavenging arrangements are classified into: (a) cross-flow (b) loop- scavenged and (c) uni-flow configurations [2]. The real scavenging process is characterized by two problems common to two-stroke engines in general: short-circuiting losses and mixing[1]. In short circuiting process, there is mixing of fresh air fuel mixture with exhaust gases resulting in nearly 35% loss of fresh charge through the exhaust valve. This is dead, loss and it should be avoided [4]. In mixing, there is a small amount of residual gas which remain trapped without being expelled, being mixed with some of the new air charge resulting in dilution [1]. This will lead to incomplete combustion emitting significant amount of particulate matter, un-burnt hydrocarbons (UBHC), carbon mono-oxide (CO) and Nitrous oxides (NOx). Conventional two stroke engines are well known to pollute badly and their future being limited due to excessive pollution [3]. Most recently, with more stringent emission norms the need to cut down both evaporative and exhaust emissions is necessitated in automobiles. The ever depleting mineral oil resources threw another challenge for

designers to go for fuel efficient engines. To bring down the evaporative emissions, the combustion process has to be improved during light load operation. With a quest to satisfy the above needs, there is a continual search for a new two stroke cycle engine with improved air flow with modified inlet and exhaust passages[5]. The focus of this design is to avoid scavenging problems of short circuiting and mixing. Figure.1.1 gives a pictorial representation of the proposed model.

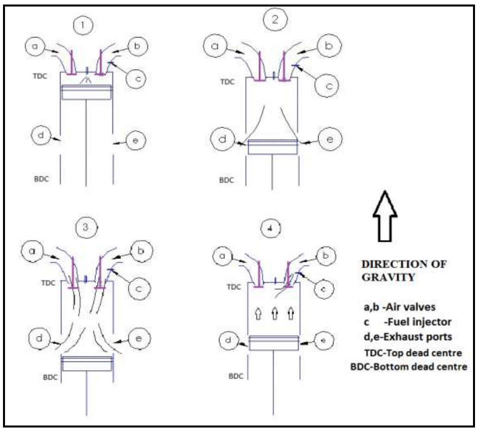

Figure 1.1: Working Model of the Proposed Engine

The working cycle of the above model is explained below

- Initially the piston is at the end of the compression stroke. Spark initiates combustion and the piston starts moving towards the bottom dead Centre.
- As the piston uncovers the exhaust ports, due to the high pressure difference between the cylinder and the exhaust manifold, the burnt gases inside the cylinder start moving out.
- When the pressure inside the cylinder drops below the pressure at inlet, the inlet air valves are opened. Due to the pressure difference the inlet air aids in pushing out the Exhaust gases.
- As the piston covers the ports the inlet air valves are closed which is followed by fuel injection. The cycle gets repeated.

2. METHODOLOGY

Four stroke IC engines use valves for intake and exhaust strokes. Changing their valve timing by cam adjustment followed by introduction of ports in the same cylinder would satisfy the structural requirements of the proposed model. Also a majority of real world two stroke engines do not have ports and valve configuration. Hence modifying them according to the proposed model would be more complex than that of a four stroke engine.

So the air flow system of a two stroke gasoline fuel injected engine is developed based on a four stroke gasoline engine. The valve timing data for four stroke Royal Enfield super meteor engine was initially considered for the two stroke engine model. The actual cycle, which took 785 degrees of cranks angle rotation was completed in 475 degrees for the two stroke model engine. The crank angle details of individual strokes were presented in TABLE 1.

Table 1: Initial Conversion of Valve Timing

Operating Stroke	Royal Enfield Super Meteor Engine(Degrees)	Proposed Model(Degrees)
Suction	270	115
Compression	120	120
Expansion	105	120
Exhaust	290	120

To obtain the air flow and other performance of the proposed engine, geometrical parameters (stroke, bore, compression ratio, valve size and timing) of Royal Enfield Super Meteor Engine were used as inputs to Diesel R K software to find the required reference engine parameters.

The compression and expansion stroke duration of the proposed model were fixed almost the same as that of the reference engine by altering the intake and exhaust stroke duration.

3. DETERMINATION OF INLET PARAMETERS

The complete combustion equation for pure octane fuel (C_8H_{18}.) is given by:

$$C_8H_{18} + 12.5\ O_2 = 8\ CO_2 + 9\ H_2O \quad (3.1)$$

The stoichiometric air to fuel ratio is 15.1:1. However, by chemical analysis for perfect gasoline/air mixture, the combustion equation becomes

$$C_8H_{15.4} + 11.85\ O_2 = 8\ CO_2 + 7.7\ H_2O \quad (3.2)$$

The stoichiometric air fuel ratio is 14.7:1. However, under wide open throttle conditions, it is considered as 14.6:1. In the present work the engine runs at a constant 5000 revolutions per minute. As no throttling is done, wide open throttle conditions can be considered for calculations. The total mass of air is found from the mass of fuel supplied and air fuel equivalence ratio as in Figure 4.1.

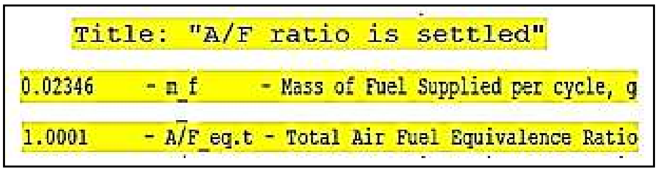

Figure 3.1: Air Fuel Ratio and Mass of Fuel Supplied in Reference Engine

Mass of Air Determination

1.001= (A/F actual) / A/F stoichiometric) (3.3)

14.6×1.001 = A/F actual Mass of fuel supplied = 0.02346×10^{-3} kg Actual air supplied = 3.425×10^{-4} kg

Total mass flow during intake = mass of air + mass of fuel

= 3.425×10^{-4} kg + 0.02346×10^{-3} kg= 3.65×10^{-4} kg

Velocity of air flow through valve orifice can be obtained by applying flow rate equation,

$\Pi/4 \times d^2 \times \rho \times$ Velocity $\times$ intake duration = 3.65×10^{-4} kg

Intake duration = 9× 10^{-3} seconds (270 degrees @ 5000 rpm) d =diameter of valve orifice (2.66 cm) Velocity = 60.513 m/s.

4. EXHAUST CALCULATIONS

The focus of exhaust calculations was to narrow down to the dimensions of exhaust port that is sufficient enough to send the mass of burnt gases out of the cylinder within 120 degrees of crank angle and simultaneously attain a cylinder pressure lesser than the inlet pressure. The pressure at the inlet was initially assumed to be atmospheric.

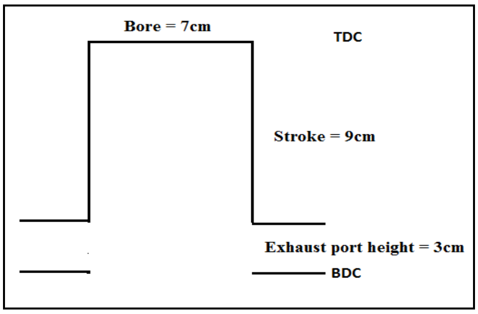

Figure 4.1: Represents the Cylinder Dimensions of the Proposed Model

The existing crankcase scavenging system engine was modified in such a way that the exhaust port opens at a piston travel of 6cms from the top dead centre. To vary the height of the exhaust port, the height of transfer port from the bottom dead centre was adjusted.. Almost the same amount of fuel (0.02350 g of gasoline) as in the reference engine was supplied. The Air fuel equivalence ratio was maintained at 1.001. The pressure inside the cylinder was obtained from the pressure crank angle diagram which was around 6 bar when the piston just uncovers the exhaust port and would not permit the transfer of charge into the cylinder.

So the transfer port location was changed to 1cm above BDC. This allowed the increase in exhaust port height by 2cm and the width of the port was extended to 20 cm. The amount of fuel supplied was 0.02336 g per cycle (slightly lesser than that of the reference engine). The pressure attained was found to be 0.83 bar which facilitated charge induction as the pressure inside the cylinder is less than atmospheric pressure. In the next iteration 0.02361 g of gasoline was supplied (slightly higher than that of the reference engine).

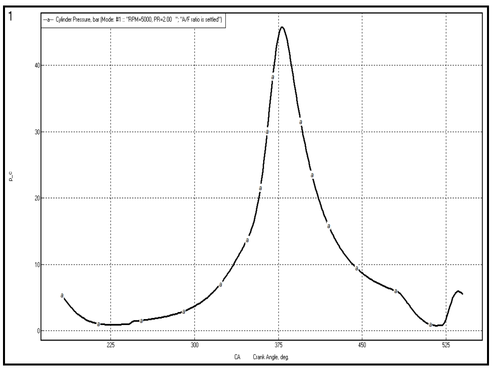

Figure 4.2: Pressure Vs Crank Angle Diagram of Crankcase Scavenged Model (0.02336 G of Gasoline)

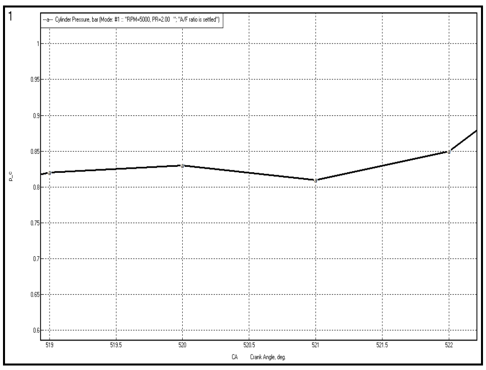

Figure 4.3: Pressure at the End of 2 Cm Height of Exhaust Port (0.02336 G of Gasoline)

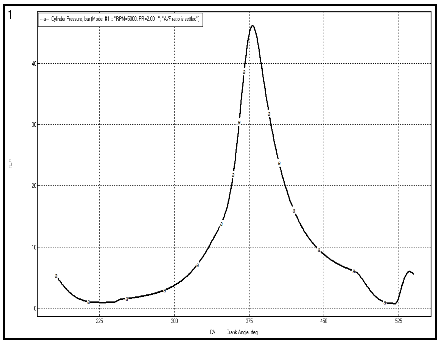

Figure 4.4: Pressure Vs Crank Diagram of Crankcase Scavenged Model (0.02361 g Fuel)

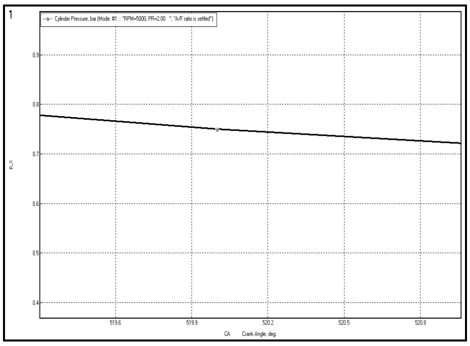

Figure 4.5 Pressure at the End of 2 Cm Height of Exhaust Port (0.02361 g Fuel)

With the port dimension changed to 2 cm height and 20 cm width the cylinder pressure is lowered below the atmospheric pressure at the end of expansion. To check whether the exhaust port dimension is sufficient enough to send out 3.65×10^{-4} kg of burnt gases, two separate integrals were done, one for velocity (Figure. 5.6) and another for area (Figure. 5.7). The pressure crank angle diagram for 0.02336 g fuel injection was chosen for calculation as it is closest to the amount of fuel injected in reference engine (0.02346 g).

The velocity is integrated for the time interval of 40 degrees. The equation of area for the exhaust port is integrated to determine the length, The peak temperature is 2545 degree Celsius and the corresponding pressure is 46.8 bar. The corresponding density of air is 0.5780 kg/m^3. The exhaust mass flow is estimated to be 3.075×10^{-4} kg.

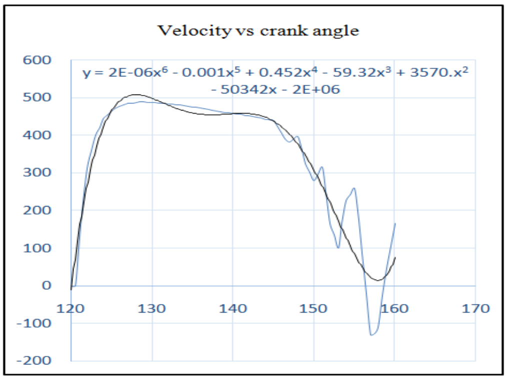

Figure 4.6: Velocity Curve (M/S) of Exhaust Gases

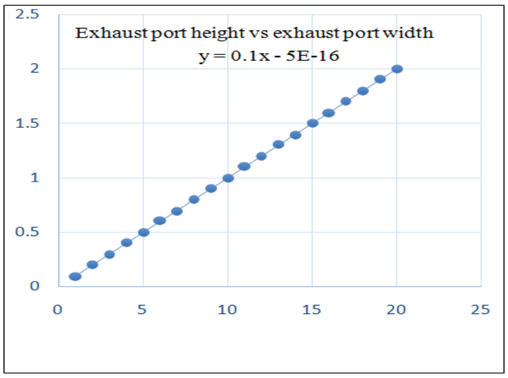

Figure 4.7: Area (Cm2) Curve of Exhaust Ports

5. INTAKE CALCULATIONS

The objective of intake design was to permit 3.65×10^{-4} kg of fresh air within 80 degree of crank angle as the desirable cylinder pressure occurs only after 40 degree of exhaust process. However the compression in cylinder starts when the first piston ring reaches point A.

$$\text{Mass of the air inducted} = (\pi / 4) \times \mathrm{d}^2 \times \rho \times V \times time \times n = 3.65 \times 10^{-4}\, Kg \quad (5.1)$$

d = 2.66 cm. The diameter of the valve orifice was fixed and the intake pressure was modified to achieve the suitable design. Figure.6.4 shows variation of valve opening area with crank angle. Forced induction at 1.3 bar inlet pressure was carried out in the first iteration.

3 valves of diameter 26.6 mm pump in $4.7 \times 10^{-4}\ Kg$ of air which was higher than the required amount of air during intake (3.65 X 10-4 kg.)The next iteration was carried out at a pressure 1.1 bar. Accounting in density variations with pressure

$$P = \rho \times R \times T \tag{5.2}$$

$1.1\times10^{-5} / (288 \times 297) = 1.303\ kg/m^3$

The mass of air through the inlet was 3.9783 X 10-4 kg which was still more than required.

Figure 5.1 and Figure 5.2 shows variation of intake pressure and intake velocity with crank angle(CA) for forced induction at 1.1 bar.

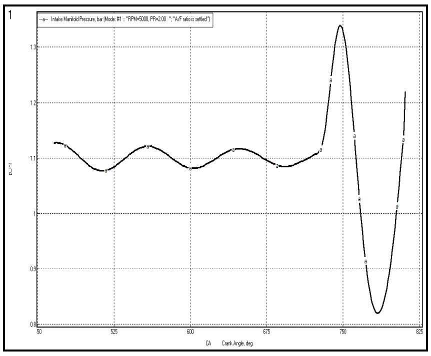

Figure 5.1: Intake Pressure Variation Vs CA at 1.1 Bar

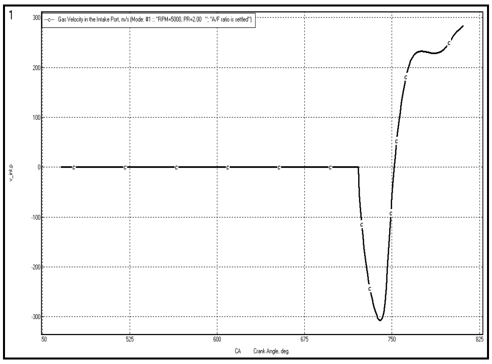

Figure 5.2: Intake Velocity Variation Vs CA at 1.1 Bar

The next iteration was carried out for natural aspiration. Mass of air through inlet was 3.6016 X 10^{-4} kg, which was slightly less than required. The cylinder pressure during the opening of inlet valve is 0.75 bar which would permit the flow of air from intake at 1 bar into the cylinder. Thus natural aspiration would serve the intake purpose. However forced induction at 1.1 bar would be the most suitable intake design.

6. EFFECTS OF GRAVITY

The piston opens, exhaust ports as it moves against gravity during the expansion stroke. This is analogous to inverting the cylinder of the conventional IC engine. A scaled model was developed and the acceleration and velocity effects, were determined after inverting it.

Table 2: Specifications of Crank Slider Components

Part Name	Height/Length(Cm)	Width/Diameter(Cm)	Thickness(Cm)
Piston	3	7	2
Connecting rod	16.5	2	2
Crank rod	4.488	2	2

Material for piston was chosen as aluminum, steel for connecting rod and crank rod. The mechanism was simulated at 5000 rpm and a force of 18000.62 N (46.8 bar peak pressure) was applied on the cylinder head for every cycle. Figure. 6.1 shows the ADAMS simulation setup.

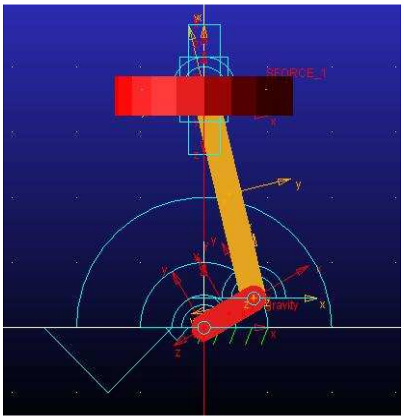

Figure 6.1: Crank Slider Mechanism Simulation Setup

The plots of velocity and acceleration versus time were unaffected by the change in gravity direction. Figure 6.2 and 6.3 represent the velocity vs time graph and acceleration vs time graph respectively.

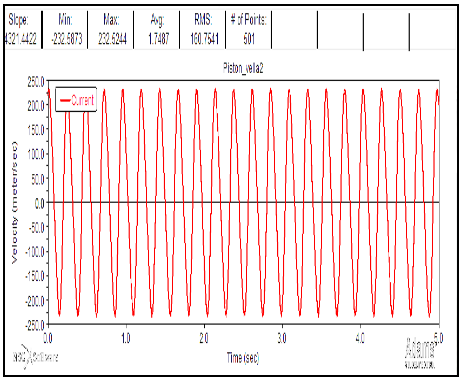

Figure 6.2: Velocity Graph For Inverted Crank Slider Mechanism

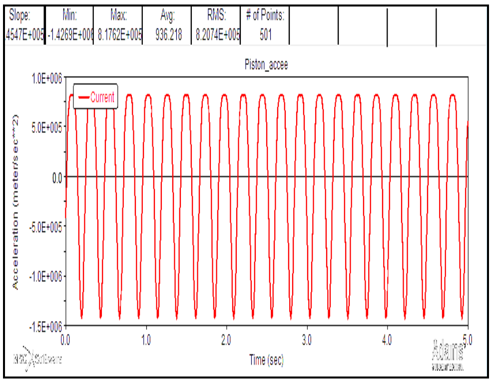

Figure 6.3: Acceleration Graph Inverted Crank Slider Mechanism

Bottom induction of fresh air is chosen for the following reasons. The intake manifold heats up the incoming air which is at a lower temperature. The temperature of exhaust gases is also very high. As a natural phenomenon, hot gases tend to move up due to lesser density. To avoid any backflow of incoming fresh air and mixing of fresh air with burnt gases in the combustion chamber, the gases are allowed to flow in the direction that is facilitated naturally.

Figure 6.4 shows the temperature distribution inside the cylinder. The peak temperature of the cylinder is 2545 (Fig. 6.5) degree Celsius and at a pressure of 46.8 bar. The density of air inside the cylinder will be lesser than0.5780 kg/m^3. The average temperature at the crankcase wall was 60 degrees Celsius. Assuming this temperature and ambient pressure of 1 bar to exist in the intake manifold, the density of fresh air will be higher than 1.060 kg/m^3 as (during the intake process, the incoming charge is usually cooler than the walls and the flow velocities are high. [2]. This gradient will force the fresh charge into the cylinder to push out the burnt gases.

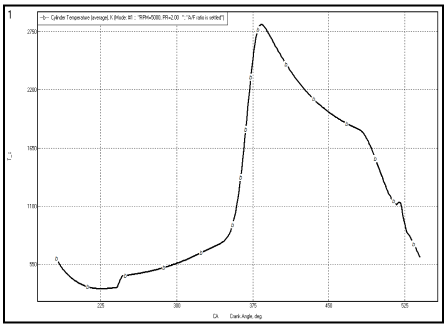

Figure 6.4: Temperature Distribution Inside the Cylinder.

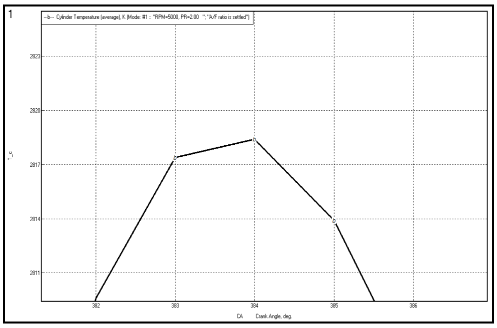

Figure 6.5: Peak Temperature Occurrence

The temperature of the exhaust manifold is 727 degree Celsius and corresponding pressure of air is 0.6 bar at the time of opening of the exhaust ports. As the gases of reduced density from combustion chamber start moving out through the exhaust port, the cylinder pressure at 0.85 bar prevents the backflow into the combustion chamber from the exhaust manifold. This process supports opening of inlet valves that permits flow of the fresh air, which is relatively at a higher density from the intake manifold of pressure 1 bar. This will avoid all possibilities of backflow of burnt gases back into the combustion chamber. The changes in velocity due to effect of gravity were examined for entry of fresh air into the cylinder and injection of fuel. The rate at which air was supplied was 0.114166 kg/Sec for 0.003 seconds (5000 RPM and 90 degree inlet). This ensures the supply of required 3.425×10^{-4} kg of air with atmospheric pressure being maintained at the exhaust port.

Figure 6.6 and Figure 6.7 shows the velocity contour for air flow downward and upward respectively.

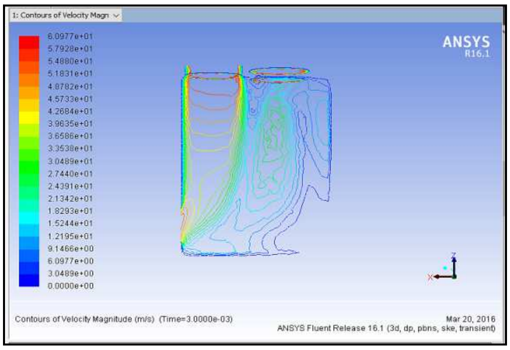

Figure 6.6: Velocity Contour for Air Flow in Downward Direction

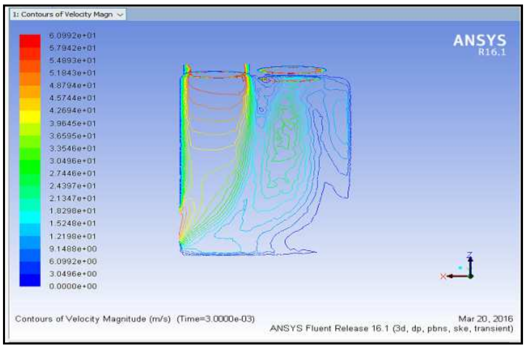

Figure 6.7: Velocity Contour for Upward Air Flow

The contours show that there is a small change in inlet velocity. Air having very less density (1.20 Kg/m^3 at 15 degree Celsius and 1 bar pressure), gravity does not have a major impact.

The direction of fuel injection was determined from studies related to fuel injection and its effect on engine operation. It is important to have less liquid fuel impingement on the piston top, because a liquid fuel film vaporizes more slowly than airborne droplets, resulting in high hydrocarbon emissions[6]. For stratified cold operation, the spray impingement on piston may yield to film formation producing large HC and soot emissions. Moreover, liquid film may get enough heat from the piston-wall to evaporate, and then decrease the air cooling effect and the compression ratio [7]. Mie scattering images show the liquid exiting the injector probe as a stream and directly impacting the piston top. Schlieren imaging was used to show the fuel vaporizing off the piston top, later in the expansion stroke and during the exhaust stroke. Emissions tests showed that the presence of liquid fuel on in-cylinder surfaces increases engine-out hydrocarbon

emissions [8]. Spray/wall interaction has a significant influence of the mixture formation process in gasoline direct injection (GDI) engines. Moreover, the fuel wall film and the resulting delayed evaporation of the liquid fuel are the main sources of soot formation in the internal combustion engines [9].

Hence, if the fuel is not fully vaporized and properly mixed with the air in the engine's cylinders during the combustion process, part of this fuel may go out of the cylinders as unburned hydrocarbons. The carbon deposits can also cause cold starting problems as during the engine warm-up, they can actually absorb some of the fuel that is supplied for combustion. It is important to atomize the fuel to improve vaporization, which can be achieved by distributing the fuel over a greater area inside the cylinder.

The direction of fuel injection is chosen with an objective to enhance distribution inside the cylinder, thereby enhancing vaporization and reducing the magnitude of deposition over piston head. This objective is satisfied by reversing the direction of injection, against gravity. The fuel has to be injected precisely at 3.5 cm from the BDC to ensure sealed combustion chamber thereby avoiding scavenging. The flow of fuel into the cylinder, without the presence of air was analyzed for $6.6\text{x}10^{-4}$ seconds at a rate of 0.035 kg/sec which equals 0.02346 kg of fuel getting injected in 20 degrees at constant 200 bar pressure in both the cases. Figure. 8.8 and Figure. 8.9 shows the velocity contour for fuel flow downwards and upwards against the direction of gravity respectively.

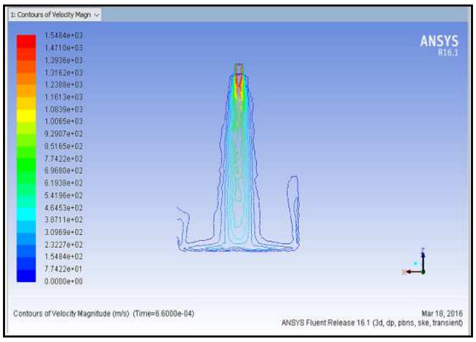

Figure 6.8: Velocity Contour for Fuel Injection along Gravity

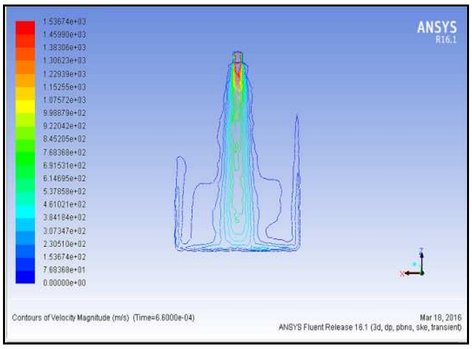

Figure 6.9: Velocity Contour for Fuel Injection against Gravity

There was 12m/s decrease in velocity (density of gasoline being 720 Kg/m^3) due to injection against the direction of gravity and the contours show that by injecting fuel against gravity the distribution inside the cylinder is improved.

7. RESULTS AND DISCUSSIONS

As obtained from the simulation results, short circuiting is avoided only when the fuel is injected in a sealed combustion chamber, to provide injection at the exact timing, and hence fuel injector is mandatory.

With the manifold design same as that of reference engine, the model engine having 90 degrees of intake stroke and forced induction at 1.1 bar and with 3 intake valves of diameter 2.66cm, is capable of sucking in the required amount of air. Forced induction is not required as the mass of charge inducted in case of natural aspiration is 3.60×10^{-4} kg instead of the required 3.65×10^{-4} kg.

The exhaust port dimensions of 2 cm height and 20 cm width is sufficient to send the exhaust out within 120 degree CA rotation with no mixing of exhaust gases with inlet fresh charge.

The possibilities of mixing of exhaust gases with inlet fresh charge has been completely eliminated with the admission of inlet air and fuel injection against gravity into the sealed combustion chamber..

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